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EXPERIMENTAL PERFORMANCE OF A 2 - 15 KILOWATT BRAYTON POWER SYSTEM IN THE SPACE POWER FACILITY USING KRYPTON

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ABSTRACT

A Brayton power system has been operated in a vacuum environment for 584 hours using krypton as the working gas. This investigation was the first operation of the complete Brayton power system. With the exception of the heat source and the heat sink (no radiator was used), the engine components represented flight-type hardware. The engine produced 9.2 kilowatts gross alternator power (7.8 kW estimated net engine power), a gross engine efficiency of 26 percent, and an estimated net engine efficiency of 22 percent at a turbine inlet temperature of 1400° F, a compressor discharge pressure of 44 psia, and a compressor inlet temperature of 80° F.

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SUMMARY

A Brayton power system has been operated in a vacuum environment for 584 hours using krypton as the working gas. This investigation was the first operation of the complete Brayton power system. With the exception of the heat source and the heat sink (no radiator was used), the engine components represented flight-type hardware. The engine produced 9.2 kilowatts gross alternator power (7.8 kW estimated net engine power), a gross engine efficiency of 26 percent, and an estimated net engine efficiency of 22 percent at a turbine inlet temperature of 1400° F, a compressor discharge pressure of 44 psia, and a compressor inlet temperature of 80° F.

INTRODUCTION

The success of space exploration depends in large measure upon the availability of electric power. The NASA Lewis Research Center is engaged in the development of various devices to produce electric power from thermal energy. One such device is the Brayton power system which operates on the Brayton thermodynamic cycle and uses an inert gas as the working fluid. The Brayton power system tested was the B type system as described in reference 1. Up to 15 kilowatts of electric power is generated by a turbine-driven alternator. A design goal for the system is 5 years of unattended operation in space. Other components of the Brayton power system include the compressor which is also driven by the turbine, the recuperator to conserve the unused thermal energy in the turbine exhaust gas, the waste heat exchanger to transfer unused thermal energy to a radiator coolant fluid, a coolant pump, a heat source, and associated engine controls. More complete descriptions of the Brayton power system may be found in the BRAYTON POWER SYSTEM DESCRIPTION section and in reference 1.

One unique feature of the engine used for the investigation was that the turbine, alternator, and compressor were mounted on a single shaft supported entirely by gas lubricated bearings. The bearings support the shaft without external pressurization except during starting and shutdown operations. The compressor and turbine have been tested separately. Results of the turbine investigation are reported in reference 2. In addition, the complete rotating package has completed 1000 hours of operation. Performance during the first 531 hours of this test is reported in reference 3.

The engine was designed to use an isotope heat source in flight applications. The isotope heat source is designed to produce turbine inlet temperatures of 1600° F. With a 25 kilowatt heat source, the engine will produce 7 kilowatts of useful electric power with a compressor outlet pressure of 32 psia. Operation of the engine with lower turbine inlet temperatures is also being considered for coupling the engine to an existing nuclear reactor. High compressor outlet pressures would be needed, however, in order to reach the desired alternator power levels. For the initial tests reported here, however, an electric heat source was used.

In a flight application, the waste heat from the engine is transferred from the gas to a liquid coolant in the waste heat exchanger. The coolant is pumped to a radiator where the unused thermal power is radiated to space. In the present investigation, the radiator was simulated with a heat exchanger which was cooled by facility refrigeration equipment.

The design working fluid for the engine is a mixture of helium and xenon gases with a molecular weight of 83.8. For the initial tests, krypton (molecular weight 83.8) was used. Although the molecular weight of krypton is the same as the design mixture, the performance of the engine would be expected to be lower due to the lower thermal conductivity of krypton.

With the exception of the working fluid, heat source, and the lack of a radiator, the engine used for this investigation was composed mostly of flight-type hardware. The objective of this investigation was to measure the performance of the engine over a range of turbine inlet temperatures, compressor discharge pressures, compressor inlet temperatures, and to demonstrate that the engine could be operated in a vacuum. To accomplish these ends, the engine was installed in the vacuum chamber of the Space Power Facility at the Lewis Research Center's Plum Brook Station. This chamber is 100 feet in diameter and was maintained at about 10^{-6} torr throughout most of the testing reported here. The engine was operated for 584 hours at turbine inlet temperatures from 1250° to 1450° F, compressor discharge pressures from 25 to 44 psia, and compressor inlet temperatures from 42° to 95° F.

BRAYTON POWER SYSTEM DESCRIPTION

The Brayton power system operates as a closed loop system and is shown schematically in figure 1. The system consists of a heat source, a power conversion system (referred to as the engine), and a heat rejection system.

Referring to figure 1, the working gas is heated by the heat source and expanded through a radial inflow turbine which in turn drives an alternator and a radial outflow compressor. After leaving the turbine, the gas flows through a recuperator where most of the energy remaining in the gas is transferred to preheat the cooler gas flowing from the compressor to the source heat exchanger. From the recuperator, the partially cooled gas passes through a waste heat exchanger where the remaining unused heat is removed. The cycle is completed as the cooled gas passes through the radial outflow compressor, the recuperator, and back to the heat source.

Power Conversion System (Engine)

The engine consists of the working gas ducting, Brayton rotating unit (BRU), Brayton heat exchanger (BHXU), electric system, gas management system, part of the heat rejection system, and associated instrumentation. Figure 2 is a picture of the engine as installed in the Space Power Facility. The engine is mounted in a frame measuring 33- by 55- by 90-inches high overall and represents a flight-type close-coupled assembly of the components.

Brayton rotating unit. - The Brayton rotating unit (BRU) is shown in figure 3. It consists basically of a radial inflow turbine (impeller diameter 4.97-in.), a four-pole brushless alternator (rotor diameter of 3.3-in.), and a radial outflow compressor (impeller diameter of 4.25-in.). All three components are mounted on a common shaft which is supported by gas lubricated journal and thrust bearings. The BRU was installed in the engine frame with the shaft vertical, turbine end up. Normal rotational speed of the shaft is 36 000 rpm with the bearings operating in a hydrodynamic (self-acting) mode lubricated by the working gas of the system. During startup or shutdown, external pressurization gas is supplied by the gas management system to the bearings to prevent bearing-to-shaft contact.

The alternator produces 120 volts (line-to-neutral), 208 (line-to-line) electric power at a frequency of 1200 Hertz. Redundant liquid cooling passages are incorporated around the alternator to carry away heat generated both by the alternator and the turbine. The BRU has a design temperature difference from turbine to compressor of about 1500° F.

The overall measurements of the BRU are 20-inches in diameter and 36-inches long. It weighs approximately 145-pounds.

Brayton heat exchanger unit. - The Brayton heat exchanger unit (BHXU) shown in figure 3 forms an assembly of the recuperator, waste heat exchanger, and ducting required to couple it to the BRU, gas management system, and heat source. Overall dimensions, less ducting, are 20-inches wide and 56-

inches long. The unit weighs approximately 440-pounds. The unit is designed to operate with a temperature difference of approximately 1150° F.

The recuperator core operates as a gas-to-gas counterflow heat exchanger. Plate- and fin-surface sandwiches are used for both gas flows and furnace-brazed into a one piece core assembly. Core dimensions are 8.5- by 20- by 20-inches.

The waste heat exchanger is constructed as a gas-to-liquid cross-counterflow unit with redundant passages on the liquid side. There are 8 liquid passes back and forth across each gas flow passage. Construction is similar to the recuperator core using plate- and fin-sandwiches brazed into a one piece core for both the gas and liquid sides. Because of the redundant liquid passages, sandwiches are stacked as active liquid, gas; inactive liquid, gas; etc. Overall core size is 6.5- by 16- by 20-inches.

The cores are assembled into the final unit by welding them to appropriate transition sections, headers, and ducts.

Gas management system. - The gas management system (GMS) supplies working gas for gas injection startups, hydrostatic support of the BRU gas bearings, operating system pressure adjustments, and gas venting. The GMS is connected to the engine by a short piece of working gas ducting (spool piece). A check valve in the spool piece, which can be remotely operated, serves to control the direction of gas flow during startup. Other components of the GMS are a high pressure (2000 psia) titanium gas storage bottle, pressure regulator, valves, filters, lines, and instrumentation. The assembled system is shown in figure 4.

Electric system. - The electric control package provides alternator field excitation, regulates alternator output voltage, controls BRU speed, and distributes alternator output power among the user's load bus, parasitic load resistor, and the dc power supply. The speed control operates to switch the parasitic load resistor across the generator bus and is designed to regulate to a speed which increases with parasitic load power. The measured speed change from near zero parasitic load power to 6 kilowatts was 36 000 rpm to 36 480 rpm.

The dc power supply contains 2 silver-cadmium batteries for system dc needs during starts and stops. It also contains a battery charger and ac rectifying circuits for engine ± 28v dc needs during normal operation.

Two static inverters are used to drive the 400 Hertz heat rejection system liquid coolant pump motors. The inverters are powered by the dc power supply. (See Figure 5.)

Operation of the Brayton power system is performed remotely through the use of an engine control and monitoring panel located in the Space Power Facility test control center. The panel is connected by cables to a signal conditioner on the engine, where signals from pressure, temperature, speed, flow, power, and other sensors are converted to 0 - 5v dc signals and transmitted to the control and monitoring panel for operator information and automatic control functions. Command and acknowledgement signals also pass through the signal conditioner. Power for the signal conditioner and control panel is provided by the dc power supply. Thus no facility power is required to control engine operation. The actual power required to operate all the engine auxiliaries at the minimum controlled speed condition was measured to be 2.0 kilowatts. A careful system load analysis indicates that approximately 600 watts of this power was used because of the particular hardware tested and was not typical of the power actually required. A base auxiliary load of 1.4 kilowatts was used to determine the estimated net engine power output for the system efficiency determination.

The control system provides the means for performing functions such as startup by gas injection, shutdown, valve operation, gas loop inventory regulation, coolant loop operation, BRU bearing external pressurization, and steady-state speed control. In addition, the control system provides protection for such things as gas loop overpressure and overcurrents. An emergency shutdown system is incorporated which can be triggered automatically by BRU overspeed or manually by the operator.

The electric control package, signal conditioner, dc power supply, inverters, and batteries are mounted on four cold plate heat exchangers. Heat generated in these packages is transferred to the heat rejection system by circulating its fluid through the cold plate.

Heat rejection system. - A silicone liquid, Dow Corning 200 is circulated through three parallel paths (fig. 1) to remove waste heat from the BHXU waste heat exchanger, cool the alternator in the BRU, and cool the electric system packages mounted on cold plates. For added reliability, two identical cooling loops are available. During normal operation, one loop is active while the other is inactive. The pump and motor in each loop are constructed as a sealed assembly.

Test Support Equipment

The Brayton engine was connected to test support equipment to charge and recover engine gas, simulate the space radiator, simulate the flight heat source, and provide instrumentation to measure the engine performance.

Gas charging and recovery. - The engine gas storage tank was charged from commercial storage bottles outside the test chamber. Suitable pumps were provided to charge the tank to 2000 psia. The gas vented from the engine was recovered in a low pressure tank and pumped back into the storage bottles for re-use.

Radiator simulator. - For test purposes, a radiator simulator heat exchanger was used in place of an actual space radiator. The facility heat sink noted consisted of a separate facility loop of Dow Corning 200 silicone liquid circulated through the radiator simulator heat exchanger by a facility pump and cooled by commercial refrigeration equipment. All the components except the radiator simulator heat exchanger were mounted on the engine frame. The radiator simulator heat exchanger was located on the railroad flatcar adjacent to the engine.

Electric heat source. - The electric heat source consisted of a central U-tube heat exchanger radiantly heated by quartz lamps mounted in modules on both sides of the heat exchanger.

Heat transferred into the working gas was controlled by varying power to the quartz lamps through a remote control panel in the control room.

Vehicle load simulator. - An electric loading device capable of switching in combinations of three-phase loads with unity, lagging, or leading power factor was used to extract power from the power generation system. The power to the vehicle load simulator was measured by a calibrated Hall effect wattmeter. For the tests reported here, only unity power factor loads were used.

Instrumentation. - The instrumentation installed for the test was divided into two categories: control and development. Control instrumentation consisted of those measurements required for the control, monitoring, and operation of the power system.

Development instrumentation consisted of sensors installed in addition to the control sensors to determine overall system performance as well as gross component performance.

The performance data presented in this report were obtained through the measurements listed in table 1. All pressure measurements were made using static pressure taps with strain gage pressure transducers connected. Total pressure probes and rakes were not required due to the low velocities in the gas loop. Temperature measurements were made using either iron-constantan or Chromel-Alumel thermocouples depending on the location. Both surface and stream (probe) types were used as required. Liquid coolant flow measurements in the heat rejection system were made using turbine-type flowmeters. Gas loop flow was measured by a short, high recovery Venturi (Dall tube).

The data were recorded using the facility digital data acquisition system.

TEST FACILITY

The test was conducted in the NASA Space Power Facility located near Sandusky, Ohio. The Space Power Facility is one of several test facilities forming the Plum Brook Station of the NASA Lewis Research Center. A 100-foot diameter by 120-foot high aluminum vacuum chamber, within a concrete enclosure, is the principle feature of this facility (fig. 6). The concrete enclosure provides shielding for the conduct of nuclear experiments as well as withstands the majority of the atmospheric pressure differential when the chamber is evacuated. Experiment access to the chamber from the adjacent assembly and disassembly areas is provided by two 50- by 50-foot doors in the chamber and enclosure, respectively.

Vacuum levels in the 10^{-6} torr range have been achieved since the facility became operational in 1969. The Brayton power system was located in the center of the 100-foot diameter chamber for testing. All performance testing was conducted under vacuum conditions and represented the first operational test program of the facility and the Brayton power system.

PROCEDURE

The operating procedures used during this investigation were established to provide for semi-automatic starting and shutdown of the engine in an effort to simulate the mode of operation expected in a flight application. In addition, several automatic functions were incorporated in the engine control system to protect the engine from potentially dangerous conditions during steady-state operation.

Starting

Prior to actual engine startup, the gas bottle in the gas management system was charged to 2000 psia with krypton gas and the engine was evacuated to a pressure of about 1 psia. The facility refrigeration equipment was turned on to provide cooling of the Dow Corning 200 coolant in the radiator simulator. Next, the engine cooling loop was started. The flow rates in the various parts of the engine cooling system were preset to design values. The electric heat source was preheated to an average tube temperature of 1500° F. At this point, the engine was ready for actual startup.

The sequence was initiated by supplying external pressurization gas to the gas lubricated bearings of the BRU to lift or "float" the bearing shoes away from the shaft. With the shaft "floating", the flow of gas within the bearings normally caused backward rotation. An indication of some (about 200 rpm) rotation confirmed to the operator that the shaft was free and he could proceed with the startup. On command from the operator, the check valve (fig. 1) was closed and the injection valve (GMS) was opened. The injection gas flowed through the engine and was discharged through the open vent valve (GMS), thus operating the engine in an open loop mode. During the injection phase of the startup, the vent valve flow remained choked to simulate a startup in space. At a preset engine speed (22 000 rpm), the injection and the vent valves were closed automatically and the check valve was allowed to open. At this point (about 15 sec after start of injection), the engine was able to accelerate itself to a rated speed of 36 000 rpm. The time history of engine speed is dependent upon many factors including the speed selected for injection cutoff, the initial temperature of the heat source, and the coolant temperature. In practice, the engine speed decreased momentarily following injection cutoff and then increased to rated speed. When the engine reached rated speed, the engine-mounted speed control applied a parasitic load to the alternator preventing an overspeed condition in the Brayton rotating unit. Excitation for the alternator was supplied from the on-board power throughout the startup. The engine control system automatically adjusted the quantity of gas in the engine to a desired value of compressor discharge pressure. Shortly after startup was completed, the external pressurization gas supply to the bearings was turned off on command from the operator and the gas bearings operated in a hydrodynamic mode. This operation was performed simultaneously on the journal and thrust bearings.

Shutdown

The shutdown procedure was initiated by simply turning off the electric heat source. When a preset turbine inlet temperature of 1100° F was reached, the external pressurization gas was supplied automatically to the bearings. Six kilowatts of parasitic load was applied on command from the operator when engine speed started to decrease. This was done to reduce the coastdown time. At about 5 rpm, the external pressurization gas to the bearings was turned off and rotation stopped. Following the actual shutdown of the gas loop rotating equipment (BRU), the heat rejection systems remained operating until all engine components were cool and the engine gas was vented.

Steady-State

The engine was operated for a total of 584 hours. This running time was accumulated in 3 periods of 92, 92, and 400 hours duration. Data are presented for a range of turbine inlet temperatures from 1250° to 1450° F, a range of compressor discharge pressures from 25 to 44 psia, and a range of compressor inlet temperatures from 42° to 95° F. The engine was allowed to stabilize for at least one hour at each condition.

The symbols used in this report are defined in Appendix A and the methods of calculation used in the data analysis are presented in Appendix B.

RESULTS AND DISCUSSION

Operating History

The Brayton engine was operated in a vacuum for 584 hours using krypton gas as the working fluid. An approximate time history of turbine inlet temperature, compressor discharge pressure, compressor inlet temperature, and gross alternator output power is presented in figure 7. The 584 hours were obtained during 3 runs of 92, 92, and 400 hours each. During the runs, two intermediate shutdowns occurred so that problems with test support equipment could be corrected. Thus a total of 5 starts and shutdowns were accumulated. Each of these starts and stops were performed as outlined in PROCEDURE. The major portion of the running time was conducted at turbine inlet temperatures between 1300° and 1400° F with compressor discharge pressures between 25 and 30 psia and a compressor inlet temperature of 80° F. Although these conditions are off design conditions, they are within the range of conditions for which the engine might be expected to be used. The highest power output from the engine was achieved for a short time at a turbine inlet temperature of 1400° F with a compressor discharge pressure of 44 psia. At this condition (224 hours, fig. 7), the engine produced 9.2 kilowatts gross alternator power (7.8 kW estimated net engine power).

The results of this first phase of engine operation have been encouraging. The rotating machinery was operated without external pressurization gas to the bearings except during starting and stopping. The recuperator and waste heat exchanger performed as expected. The gas management system was able to supply gas to the bearings and the injection flow required during starting. The heat rejection system performed as expected. The engine control system was able to start, stop, and monitor the engine operation without any major difficulties. The engine was operated on internal power during the test; but, for the purpose of these tests, a facility power supply was connected to the dc bus in order to insure that nearly zero current would flow between the on-board power supply and the batteries. This allowed the batteries to be "floated" on the bus to protect the engine from a power failure.

Engine Performance

There are three independent variables affecting the performance of the Brayton engine: turbine inlet temperature, compressor discharge pressure, and compressor inlet temperature. Several excursions of these variables were made to map the performance of the engine using krypton gas as the working fluid.

Effect of compressor discharge pressure. - The effect of compressor discharge pressure on the performance of the engine is presented in figures 8 to 12. These data were obtained with a turbine inlet temperature of 1400° F and a compressor inlet temperature of 80° F.

The compressor pressure ratio (fig. 8) is a significant parameter in determining the operation of the engine. The data of figure 8 indicate that the compressor pressure ratio was nearly constant (1.90 to 1.89) over the range of compressor outlet pressures investigated. Since the compressor operated at a constant aerodynamic speed $\left(\frac{N}{\sqrt{\theta}}\right)$ and a nearly constant pressure ratio, the aerodynamic gas flow rate $\left(\frac{W_G\sqrt{\theta}}{\delta}\right)$ would also be expected to be constant.

The primary effect of changing compressor discharge pressure then is to change the compressor inlet pressure which in turn changes the actual gas flow rate through the engine. The measured gas flow rate is presented in figure 8 as a function of compressor discharge pressure. As compressor discharge pressure increased from 25 to 44 psia, the gas flow rate increased from 0.8 pound per second to 1.34 pounds per second.

The pressures measured at various stations are presented in figure 9 as functions of compressor discharge pressure. The pressures within the engine were observed to increase linearly with compressor discharge pressure.

The pressure losses in the hot and cold sides of the recuperator were approximately 0.4 psia on each side over the range of data. The pressure loss in the gas side of the waste heat exchanger was also 0.4 psia. The pressure loss in the heat source (recuperator cold side outlet to turbine inlet) increased from 1.2 psi to 1.8 psi over the range investigated. The turbine pressure drop increased from 10.8 to 17.8 psi as compressor discharge pressure increased from 25 to 44 psia. Although the pressure drop across the turbine increased, the turbine pressure ratio (fig. 8) remained nearly constant as compressor discharge pressure was increased. This behavior would be expected since the compressor pressure ratio was nearly constant and the pressure drops in the heat transfer components of the engine were very small.

The temperatures at various stations in the engine are shown in figure 10 as functions of compressor discharge pressure. Turbine inlet and compressor inlet temperatures are independent variables. At

1400° F and 80° F, respectively, the turbine outlet temperature was 1070° F; the recuperator cold side outlet temperature was 995° F; and the recuperator cold side inlet temperature was 285° F. The fact that the turbine and compressor outlet temperatures were found to be constant over the range investigated is not surprising because the compressor and turbine operate at constant pressure ratios and speeds.

The powers developed and heat transferred in the engine are presented in figure 11. Because of the increase in gas flow rate with increasing compressor discharge pressure mentioned previously, the various heat flow rates, component powers, and gross alternator power increased. At a compressor discharge pressure of 44 psia, for example, the heat added to the gas by the heat source was 35 kilowatts and the heat rejected in the waste heat exchanger was 23 kilowatts. The difference between heat added and heat rejected (12 kW) should equal the alternator gross power output (9.2 kW) plus the heat losses from the engine. The engine heat losses consist of the heat lost from the alternator and the heat lost from the surface of the engine by radiation. The heat loss measured from the alternator cooling system was found to be 2 kilowatts at a compressor discharge pressure of 44 psia. Thus the apparent heat loss by radiation from the engine components was only about 1 kilowatt. At the same condition, the actual turbine power was 28 kilowatts and the compressor power was 17 kilowatts which leaves 11 kilowatts for the gross alternator output and the losses from the alternator. The measured gross alternator power was 9.2 kilowatts; the estimated net engine power output was 7.8 kilowatts; and the heat added to the alternator coolant was 2 kilowatts. It is interesting to note that the heat transferred in the recuperator was 59 kilowatts at a compressor discharge pressure of 44 psia. This amounts to about 60 percent of the total heat added to the gas by the recuperator and electric heat source.

The efficiencies of the various components of the engine, gross engine efficiency, and estimated net engine efficiency are presented in figure 12. The efficiencies were found to be relatively constant over the range of the data. The compressor efficiency was 77 percent; the turbine efficiency was 90 percent; and the recuperator effectiveness was approximately 90 percent. A higher recuperator effectiveness would be expected when the design mixture of helium and xenon gases is used instead of the krypton gas used for this investigation because of the higher thermal conductivity of the design mixture. The gross engine efficiency was found to be 26 percent, while the estimated net engine efficiency was 22 percent.

Effect of turbine inlet temperature. - The effect of turbine inlet temperature on the gross alternator power, estimated net engine power, gross engine efficiency, and estimated net engine efficiency is presented in figure 13. These data were obtained at a compressor discharge pressure of 29.5 psia and a compressor inlet temperature of 80° F. Gross alternator power increased from 4 kilowatts to 5.6 kilowatts as turbine inlet temperature was increased from 1225° to 1400° F. The gross engine efficiency increased from 21 to 25 percent and the estimated net engine efficiency increased from 13 to 19 percent due to the increase in engine temperature ratio.

Effect of compressor inlet temperature. - The effect of compressor inlet temperature on the gross alternator power, estimated net engine power, gross engine efficiency, and estimated net engine efficiency is presented in figure 14. These data were obtained at a compressor discharge pressure of 24.8 psia and a turbine inlet temperature of 1290° F. The gross alternator power decreased from 5 kilowatts to 3.1 kilowatts as the compressor inlet temperature was increased from 42° to 95° F. For the same range of data, the gross engine efficiency decreased from 25 percent to 19 percent and the estimated net engine efficiency decreased from 18 percent to 10.4 percent. Although more power and higher efficiencies result from lower compressor inlet temperatures, significant increases in radiator area would be required for a flight application.

SUMMARY OF RESULTS

A Brayton power system has been operated in a vacuum environment for 584 hours using krypton gas as the working fluid. Most of the operation was conducted at turbine inlet temperatures between 1300° and 1400° F, compressor discharge pressures between 25 and 30 psia, and a compressor inlet temperature of 80° F.

With the exception of the heat source and the heat sink (no radiator was used), most engine components represented flight-type hardware. There were no working fluid leaks in either the engine gas loop or the liquid cooling loop and there was sufficient heat transfer (radiation and conduction) to maintain a satisfactory operating temperature for temperature-sensitive engine equipment while running in the vacuum environment. No major difficulties with the engine hardware were encountered.

The engine produced 9.2 kilowatts gross alternator power (7.8 kW estimated net engine power), an estimated net engine efficiency of 22 percent, and a gross engine efficiency of 26 percent at a turbine inlet temperature of 1400° F, a compressor discharge pressure of 44 psia, and a compressor inlet temperature of 80° F.

APPENDIX A

SYMBOLS

A area, ft^2

C speed of sound in the gas based on static temperature

C_D discharge coefficient for Dall tube

C_p constant pressure specific heat, Btu/lb mass $^{\circ}R$

h_w pressure differential, in. of water

M a Mach number

N speed, rpm

p static pressure, lb force/$in.^2$

P power, kW

p' total pressure, lb force/$in.^2$

Q thermal power, kW

R gas constant, (ft)(lb force)/$^{\circ}R$ (lb mass)

T temperature, $^{\circ}R$

V gas velocity, ft/sec

W_C coolant flow, lb mass/sec

W_G gas loop flow rate, lb mass/sec

γ ratio of specific heats

Δ difference operator

δ pressure correction to standard conditions

ϵ heat transfer effectiveness

η efficiency

θ temperature correction to standard conditions

ρ density, lb mass/ft^3

% percent

Subscripts:

a	alternator
AD	added
c	compressor
C	coolant
G	gas
GR	gross
n	net
r	recuperator
t	turbine
w	waste
1	compressor inlet condition
2	compressor outlet condition
4	recuperator cold side inlet condition
5	recuperator cold side outlet condition
6	Dall tube
10	turbine inlet condition
11	turbine outlet condition
12	hot side recuperator inlet condition
37	waste heat exchanger coolant inlet condition
38	waste heat exchanger coolant outlet condition
41	alternator coolant inlet condition
42	alternator coolant outlet condition

APPENDIX B

METHODS OF CALCULATION

The following equations and definitions, along with the ideal gas relationships, were used to calculate the performance of the engine from the measured data.

The gas mass flow rate was calculated by using the following equation:

$$W_G = 1.5318(C_D)(P_5 - 0.009595\, h_w)(0.9905151 + 1.90839\, T_5\, 10^{-5})\sqrt{\frac{h_w}{P_5\, T_5}} \qquad \text{(B1)}$$

where

$$h_w = (P_6)\,(27.832) \qquad \text{(B2)}$$

During the calibration of the Dall tube, the discharge coefficient was obtained as a function of the gas Reynolds number. For a first attempt to calculate the flow rate, a Reynolds number and discharge coefficient were assumed. Then with this calculated flow rate, a check of the assumed Reynolds number was made and if the two disagreed, another Reynolds number was assumed and the same process was repeated. When the difference between the assumed Reynolds number and the calculated Reynolds number was less than 1 percent, the flow rate value was within the accuracy of the measurements used to obtain it.

Total to static pressure ratio was calculated as follows:

$$\frac{p'}{p} = \left[1 + \frac{\gamma - 1}{2} M^2\right]^{\frac{\gamma}{\gamma - 1}} \qquad \text{(B3)}$$

where

$$M = \frac{V}{C}$$

$$V = \frac{W_G}{A\rho} \qquad \text{(B4)}$$

$$C = \sqrt{RT\gamma\,(32.17)}$$

The difference between total and static temperature is neglected in equations (B3) and (B4) because it was found to be very small.

Total pressure was calculated as follows:

$$p' = \left(\frac{p'}{p}\right) p \tag{B5}$$

where p is measured

Actual turbine power was calculated as follows:

$$P_t = W_G C_{P_G} \ (T_{10} - T_{11}) \ (1.055) \tag{B6}$$

Ideal turbine power was calculated by:

$$\text{Ideal } P_t = T_{10} W_G C_{P_G} \left[1 - \left(\frac{p'_{11}}{p'_{10}}\right)^{\frac{\gamma - 1}{\gamma}} \right] (1.055) \tag{B7}$$

The turbine efficiency was calculated as the ratio of actual turbine power to ideal turbine power.

Actual compressor power was calculated as follows:

$$P_c = W_G C_{P_G} \ (T_2 - T_1)(1.055) \tag{B8}$$

The ideal compressor power based on inlet and outlet total pressures was calculated as follows:

$$\text{Ideal } P_c = T_1 W_G C_{P_G} \left[\left(\frac{p'_2}{p'_1}\right)^{\frac{\gamma - 1}{\gamma}} - 1 \right] (1.055) \tag{B9}$$

The compressor efficiency was calculated as the ratio of ideal compressor power to actual compressor power.

The recuperator effectiveness was calculated as follows:

$$\text{Cold Side } \epsilon_r = \frac{T_5 - T_4}{T_{12} - T_4} \tag{B10}$$

Thermal power transferred from the cold side of the recuperator was calculated as follows:

$$Q_r = W_G C_{P_G} (T_5 - T_4)(1.055) \tag{B11}$$

Alternator cooling was calculated as follows:

$$Q_a = W_{C_a} C_{P_C} (T_{42} - T_{41})(1.055) \tag{B12}$$

where W_{C_a} was measured with a turbine type flowmeter

Thermal power transferred to the gas from recuperator outlet to turbine inlet was calculated as follows:

$$Q_{AD} = W_G C_{P_G} (T_{10} - T_5)(1.055) \tag{B13}$$

The estimated net engine output power was calculated as follows:

$$P_n = P_{a_{GR}} - 1.4 \text{ kW} \tag{B14}$$

where 1.4 kW represents the estimated engine internal power requirement

The gross engine efficiency was calculated as follows:

$$\eta_{GR} = \frac{P_{a_{GR}}}{Q_{AD}} \tag{B15}$$

The estimated net engine efficiency was calculated as follows:

$$\eta_n = \frac{P_n}{Q_{AD}} \tag{B16}$$

The thermal power rejected from the engine in the liquid side waste heat exchanger was calculated as follows:

$$Q_w = W_C C_{P_C} (38T - 37T)(1.055) \tag{B17}$$

REFERENCES

1. Brown, William J.: Brayton B Power System - a Progress Report. Paper presented at the Fourth Intersociety Energy Conversion Engineering Conf., Sept. 21-26, 1969.

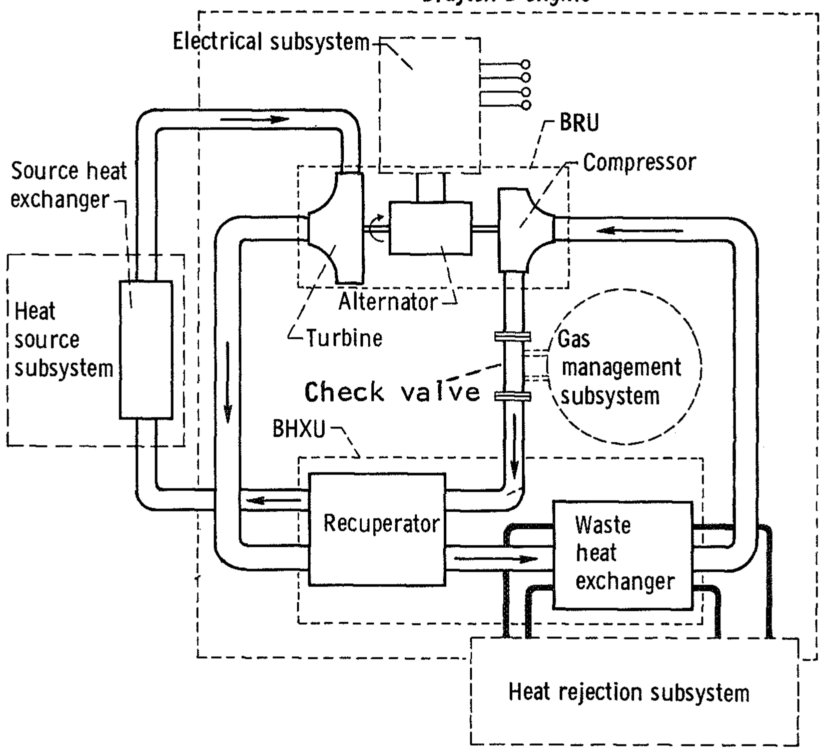
2. Nusbaum, William J.; Kofskey, Milton G.: Cold Performance Evaluation of 4.97-Inch Radial-Inflow Turbine Designed for Single-Shaft Brayton Cycle Space-Power System. NASA TN D-5090, 1969.

3. Wong, Robert Y.; Klassen, Hugh A.; Evans, Robert C.; and Wincig, Charles H.: Preliminary Investigation of a Single-Shaft Brayton Rotating Unit Designed for a 2- to 10-Kilowatt Space Power Generation System. NASA TM X-1869, September 1969.

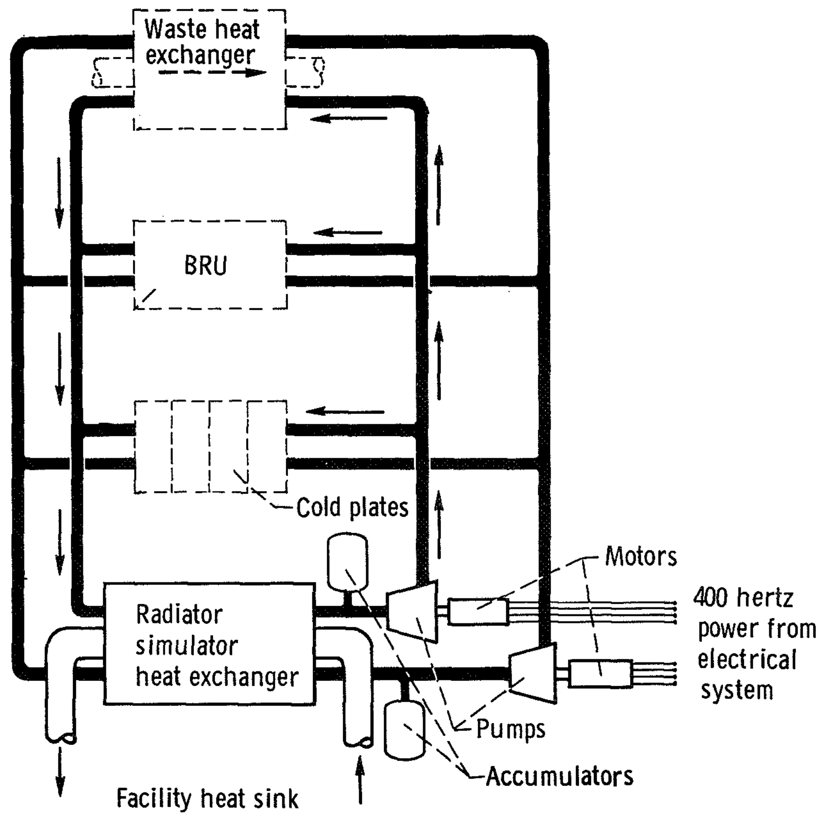
TABLE I

INSTRUMENTATION USED FOR PERFORMANCE EVALUATION

SENSOR LOCATION AND MEASUREMENT	SENSOR QUANTITY	SENSOR TYPE
Compressor Inlet/Waste Heat Exchanger Outlet		
Temperature	2	Probe Thermocouple (Type I-C)
Pressure	1	Strain Gage Transducer
Compressor Outlet Pressure	1	Strain Gage Transducer
Compressor Outlet/Recuperator Cold Side Inlet		
Temperature	3	Probe Thermocouple (Type C-A)
Recuperator Cold Side Inlet		
Pressure	1	Strain Gage Transducer
Recuperator Cold Side Outlet		
Temperature	1	Probe Thermocouple (Type C-A)
Pressure	2	Strain Gage Transducer
Turbine Inlet Temperature	3	Probe Thermocouple (Type C-A)
Pressure	2	Strain Gage Transducer
Turbine Outlet/Recuperator Hot Side Inlet		
Temperature	3	Probe Thermocouple (Type C-A)
Pressure	1	Strain Gage Transducer
Waste Heat Exchanger Inlet (Coolant)		
Temperature	3	Surface Thermocouple (Type I-C)
Pressure	1	Strain Gage Transducer
Weight Flow	1	Turbine Type Flowmeter
Waste Heat Exchanger Outlet (Coolant) Temperature	3	Surface Thermocouple (Type I-C)
Alternator Coolant Inlet		
Temperature	1	Surface Thermocouple (Type I-C)
Pressure	1	Strain Gage Transducer
Flow	1	Turbine Type Flowmeter
Alternator Coolant Outlet		
Temperature	1	Surface Thermocouple (Type I-C)
Gas Flow (Dall Tube)		
Pressure Differential	2	ΔP Strain Gage Transducer
Alternator		
Gross Power (3 Phase)	1	Solid State Quarter Square Multiplier
Vehicle Load Power	3	Hall Effect Wattmeters with Output From Each Phase Summed for Total Power
Rotating Unit Speed		
Speed	2	Capacitance Probe, Counting Pulses per Revolution



Schematic diagram, Brayton power system.



Heat rejection subsystem schematic diagram.

Figure 1. Schematic diagram, Brayton power system and Heat rejection subsystem schematic diagram

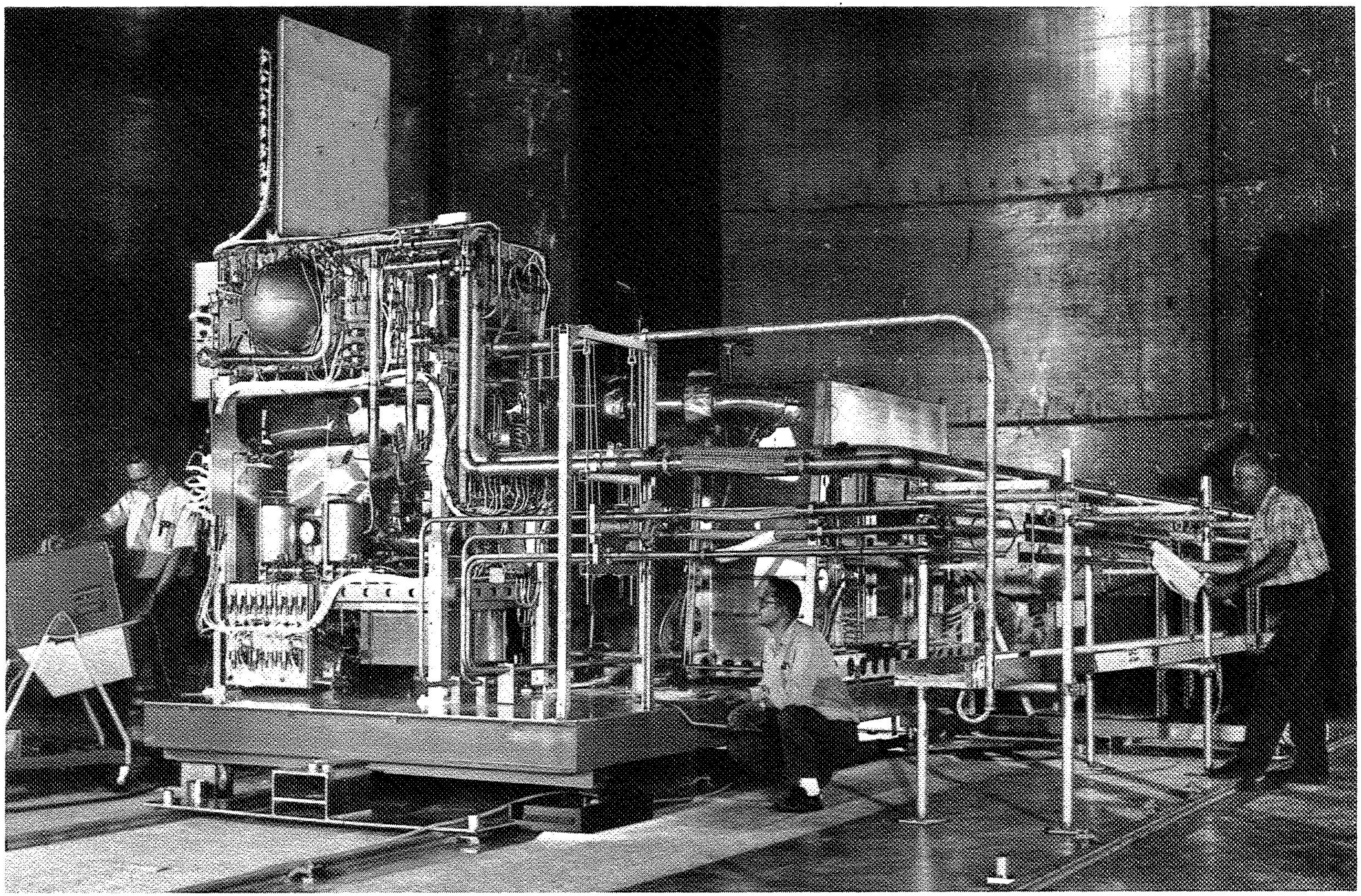
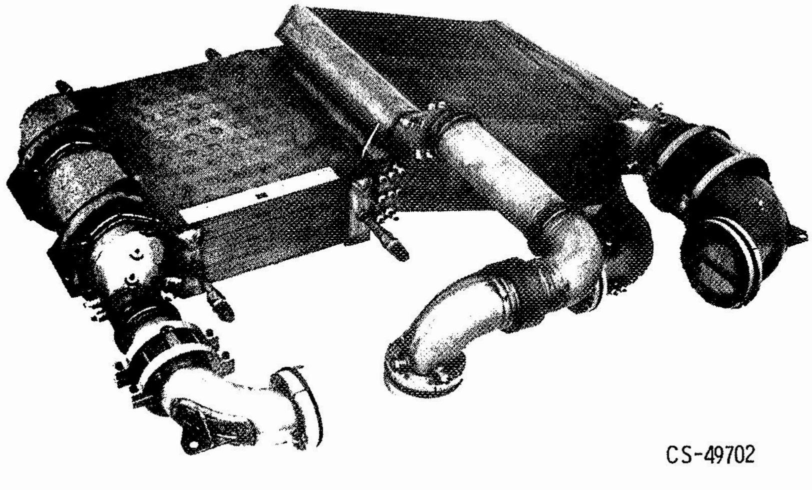


Figure 2. Picture of Brayton engine as installed in SPF

E-5510

CS-49701

Brayton rotating unit



CS-49702

Brayton heat exchanger unit

Figure 3 - Brayton Rotating Unit (BRU) and
Brayton Heat Exchanger Unit (BHXU)

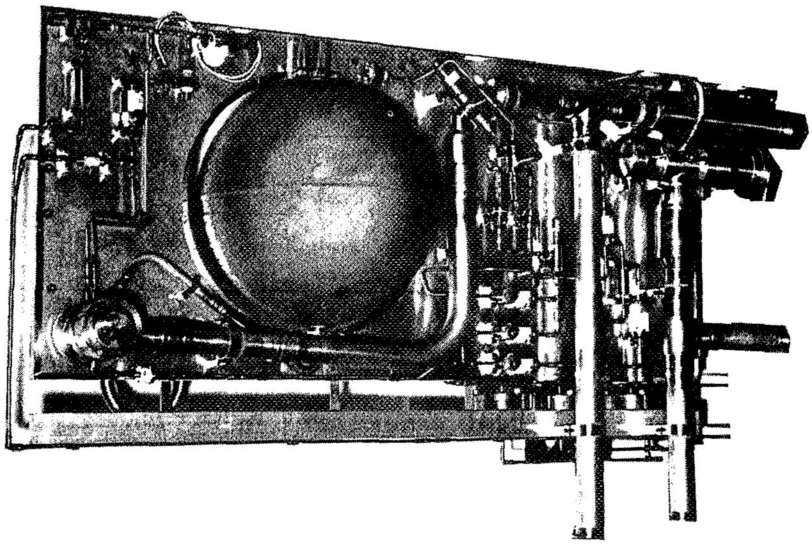


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Figure 4 Brayton gas management system.

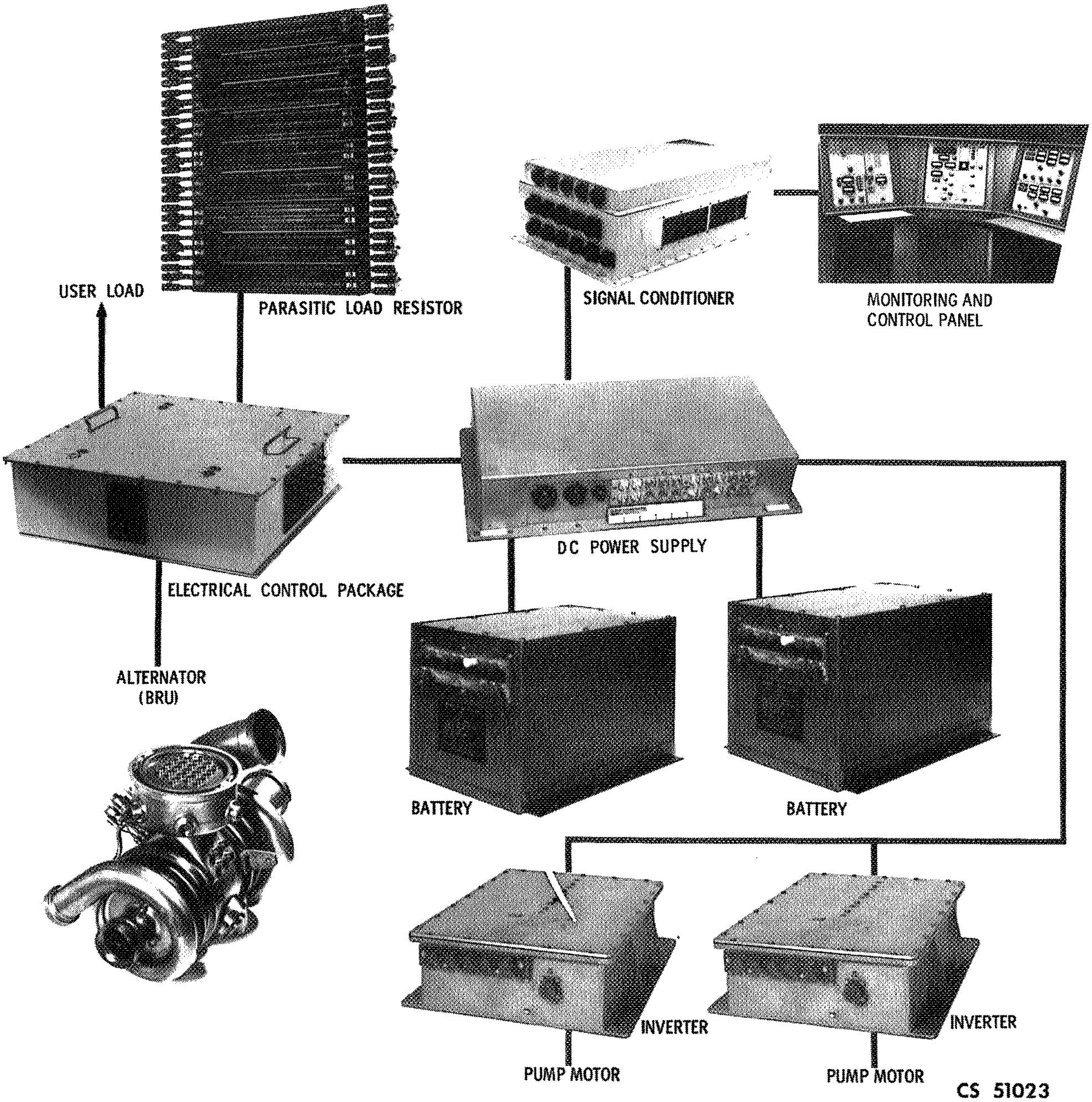

Figure 5 - Electrical System Components

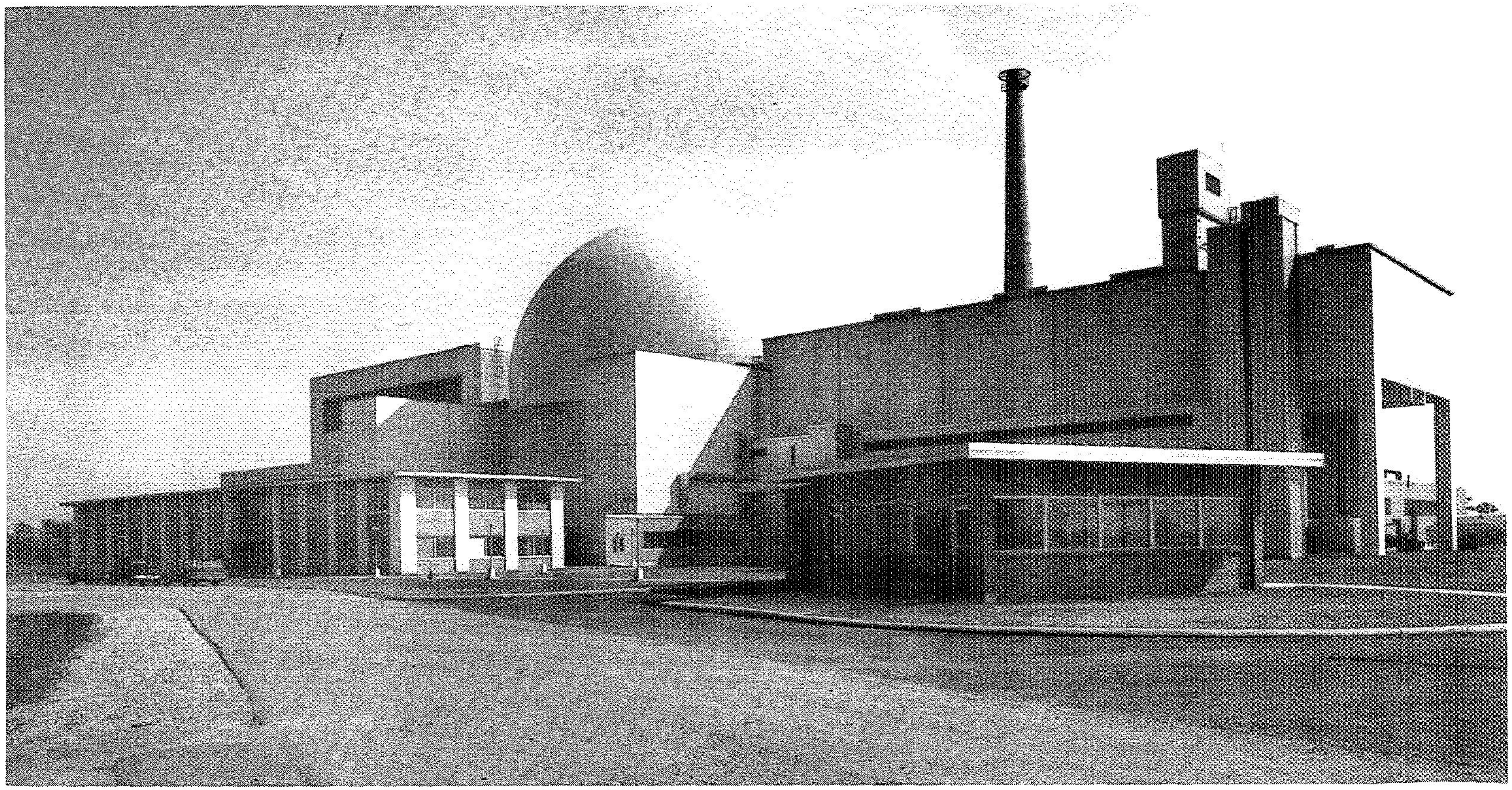

Figure 6a. Photograph of Space Power Facility

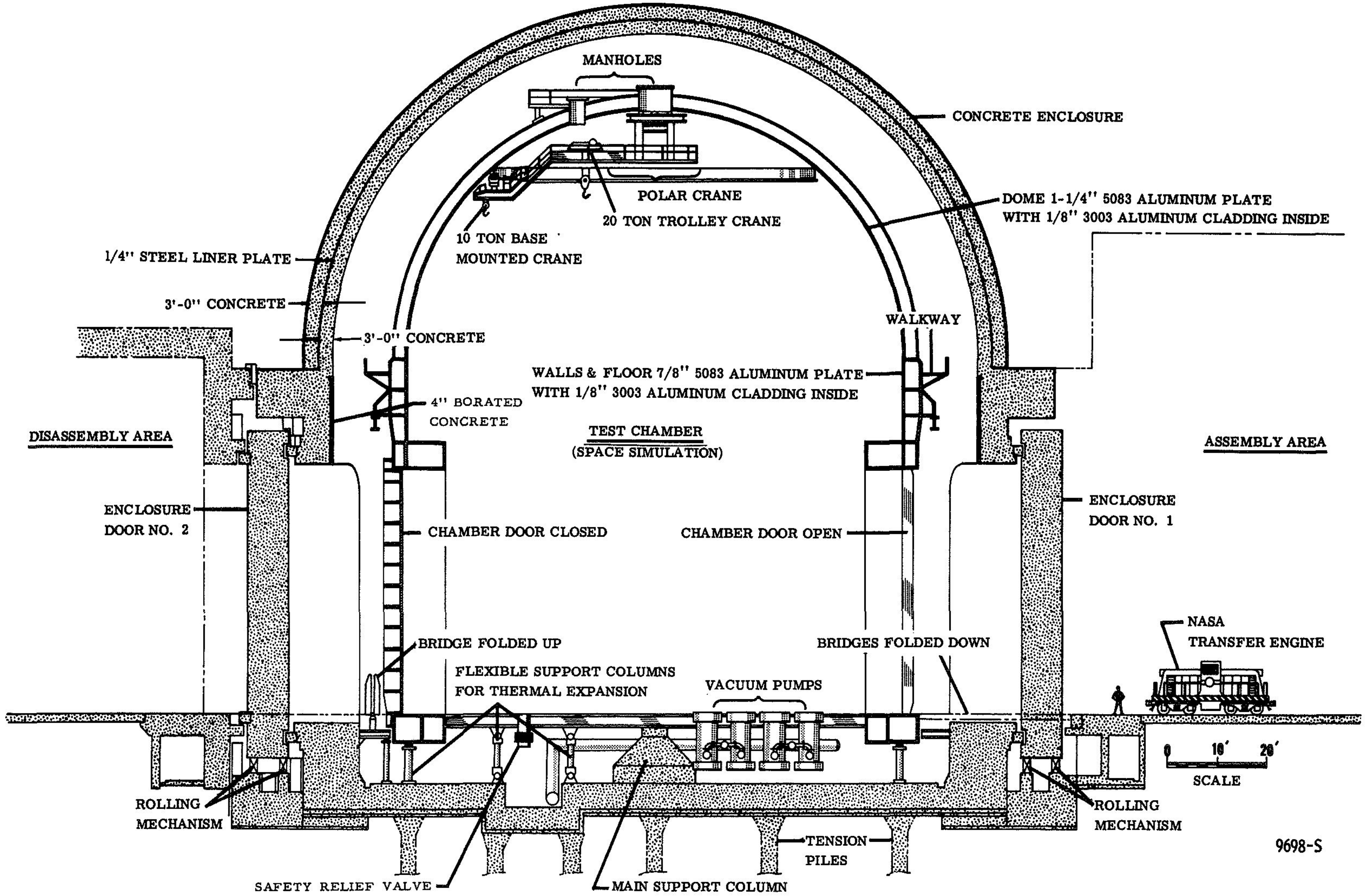

Figure 6b - Cross section through Test Chamber - Space Power Facility

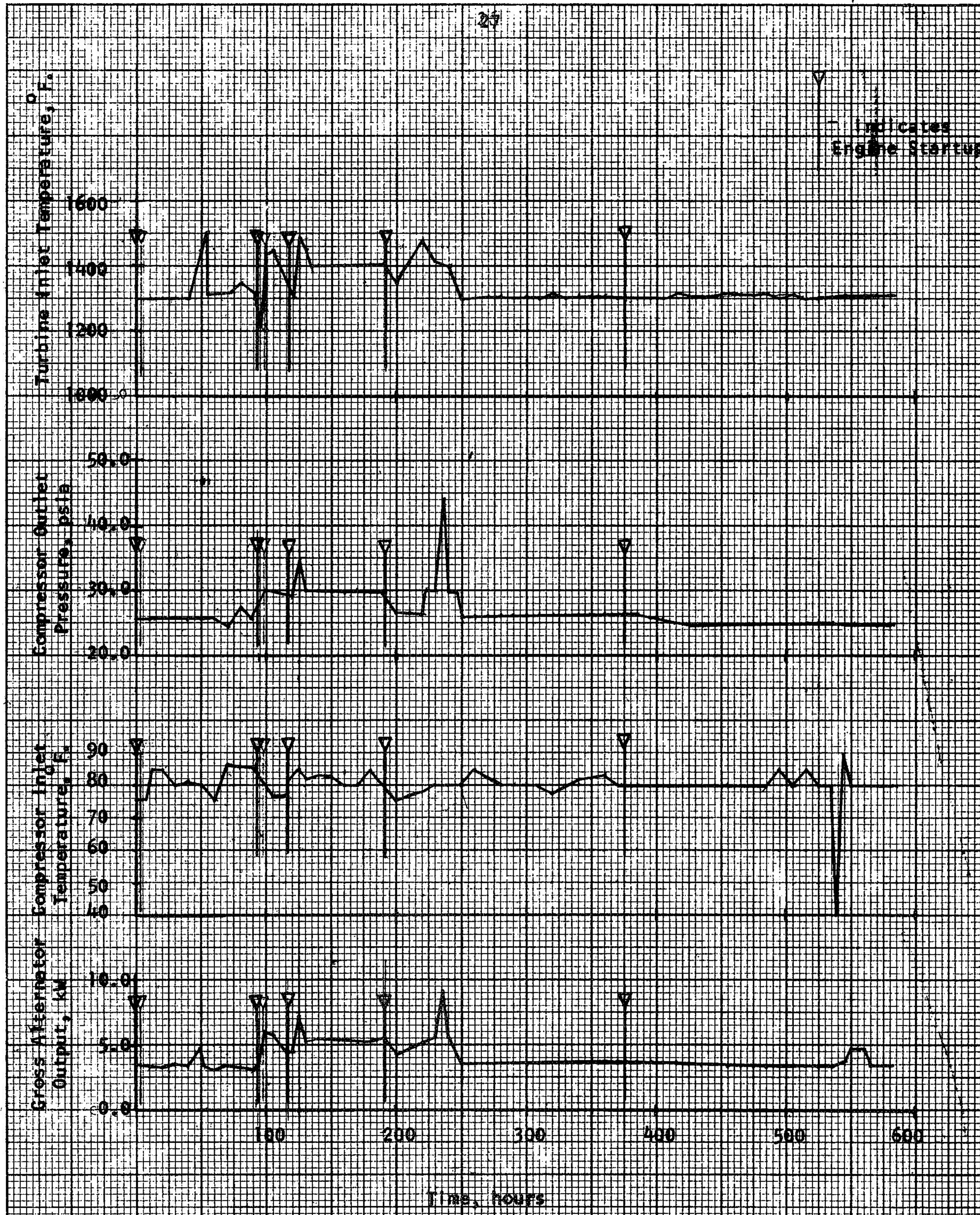

Figure 7. Engine Variables vs. Time

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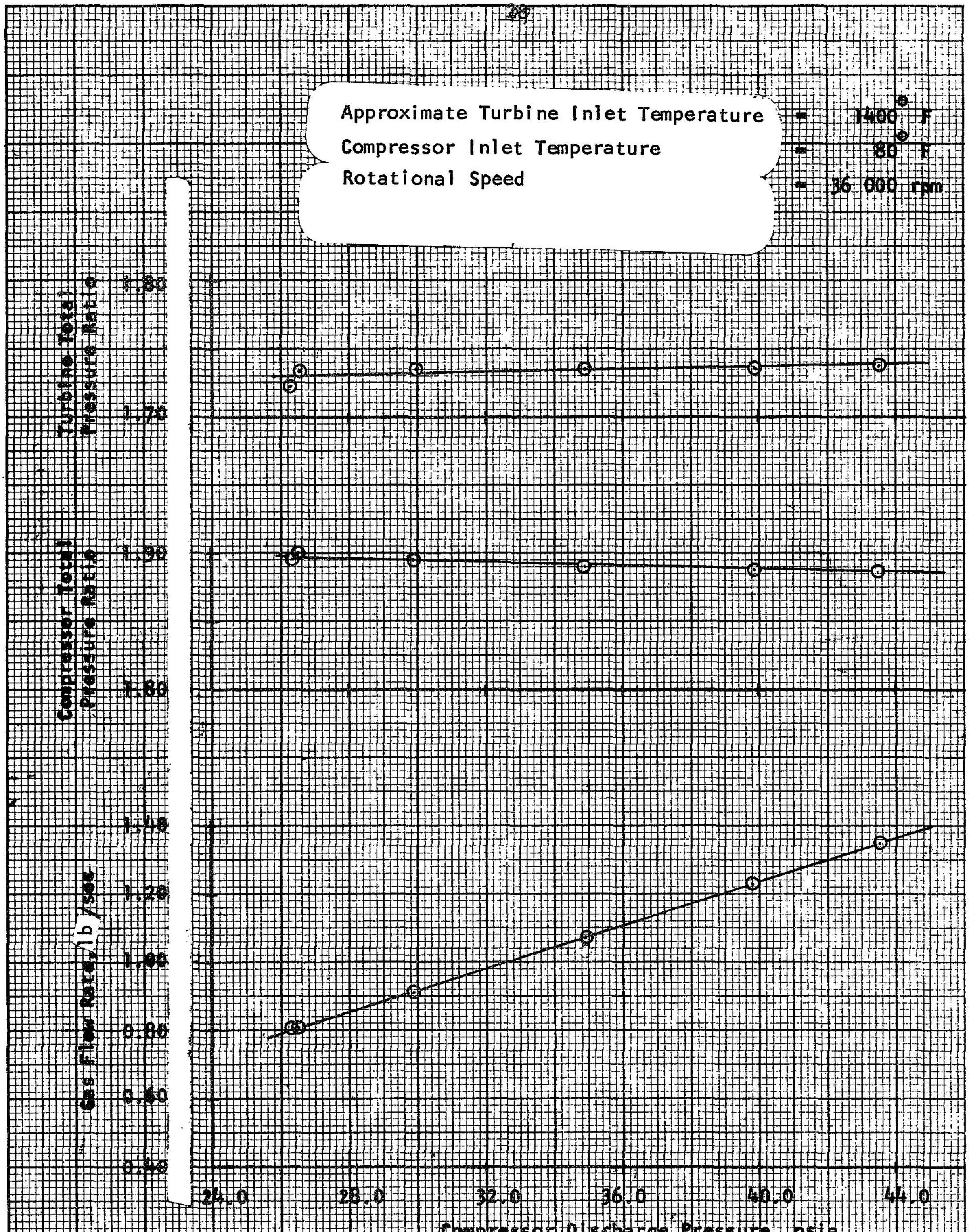

Figure 8. Variation of Gas Mass Flow Rate and Turbomachinery Pressure Ratios with Compressor Discharge Pressure

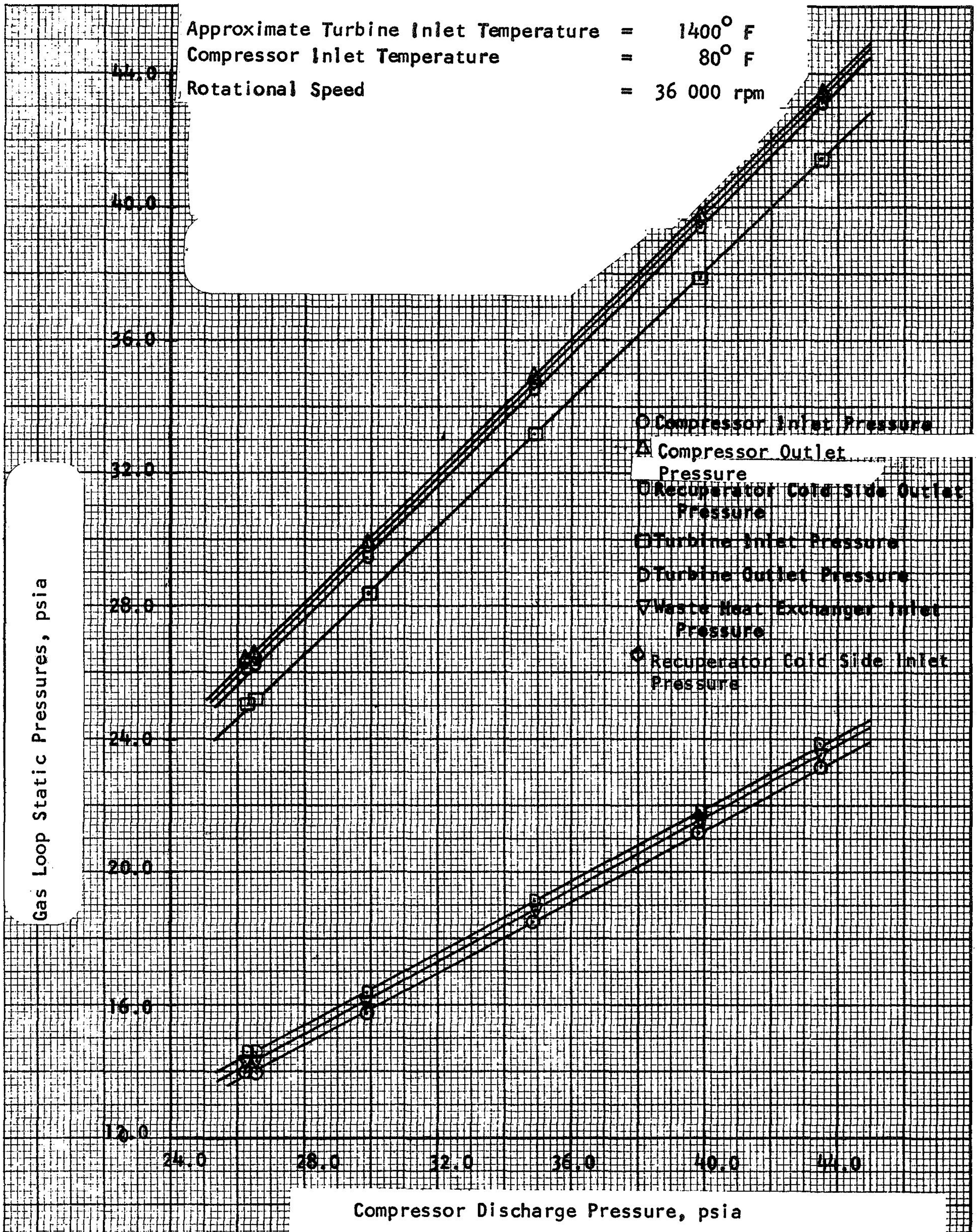

Figure 9. Variation of Gas Loop Static Pressures with Compressor Discharge Pressure

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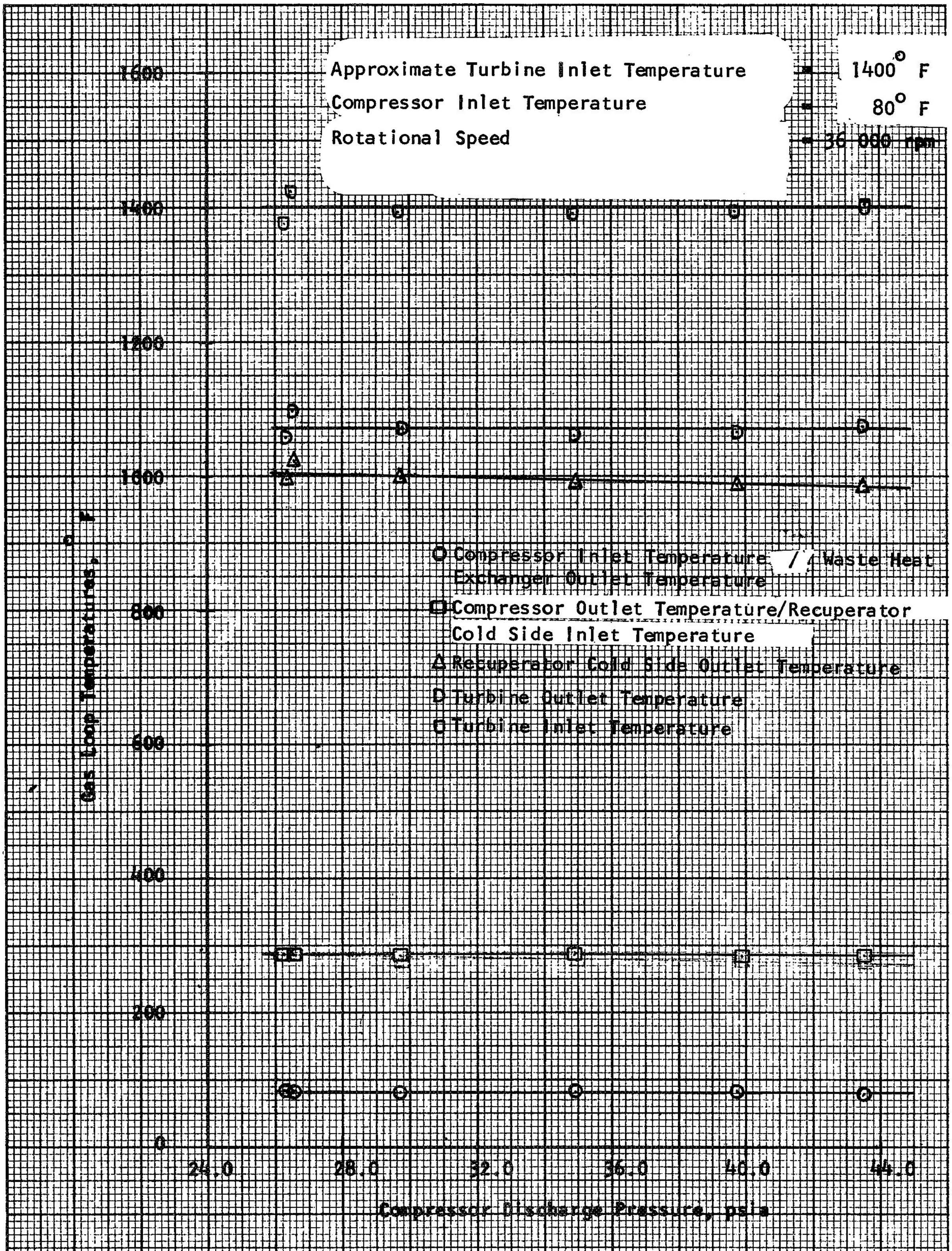

Figure 10. Variation of Gas Loop Temperatures with Compressor Outlet Pressure

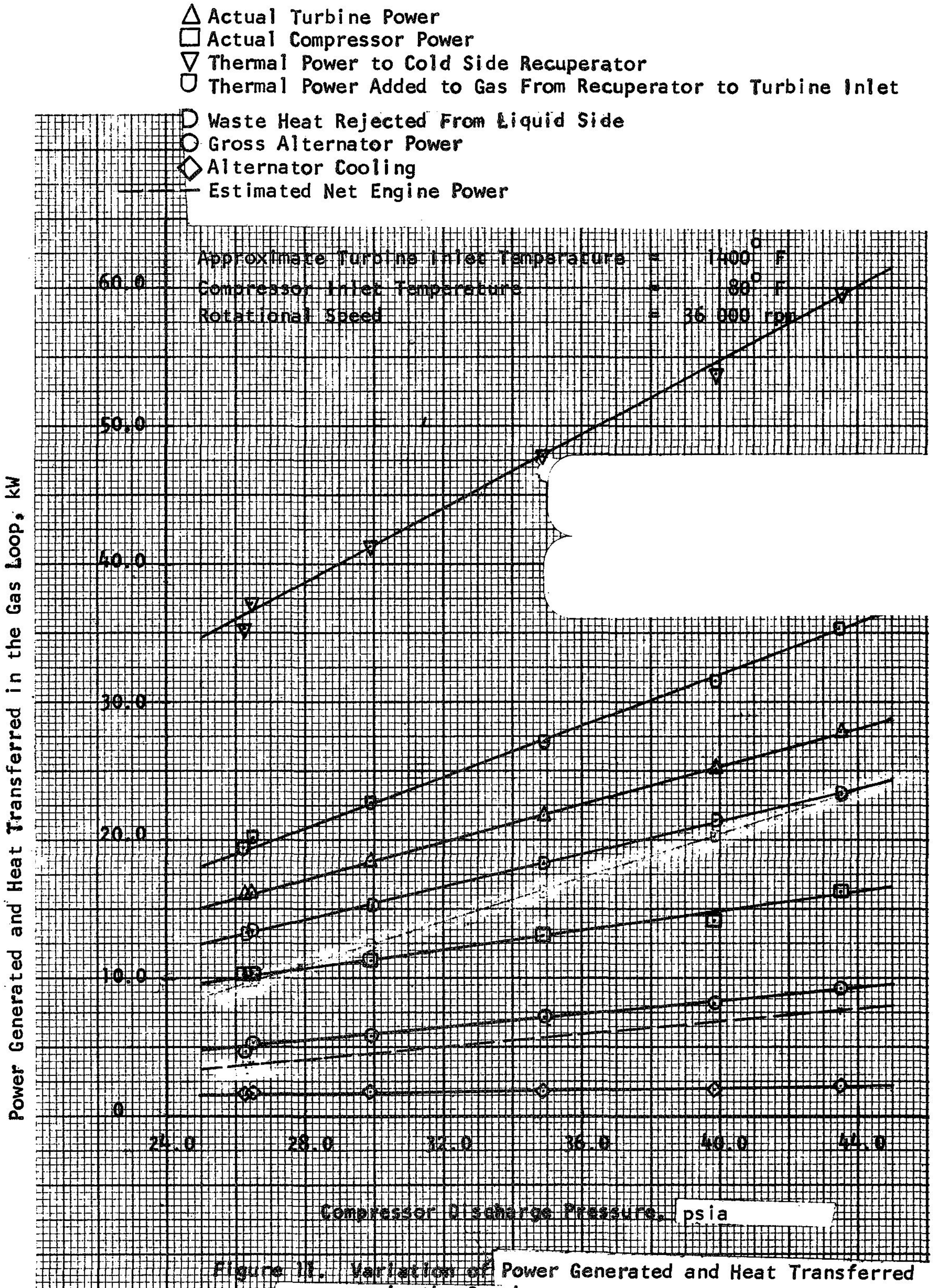

Figure 11. Variation of Power Generated and Heat Transferred in the Gas Loop

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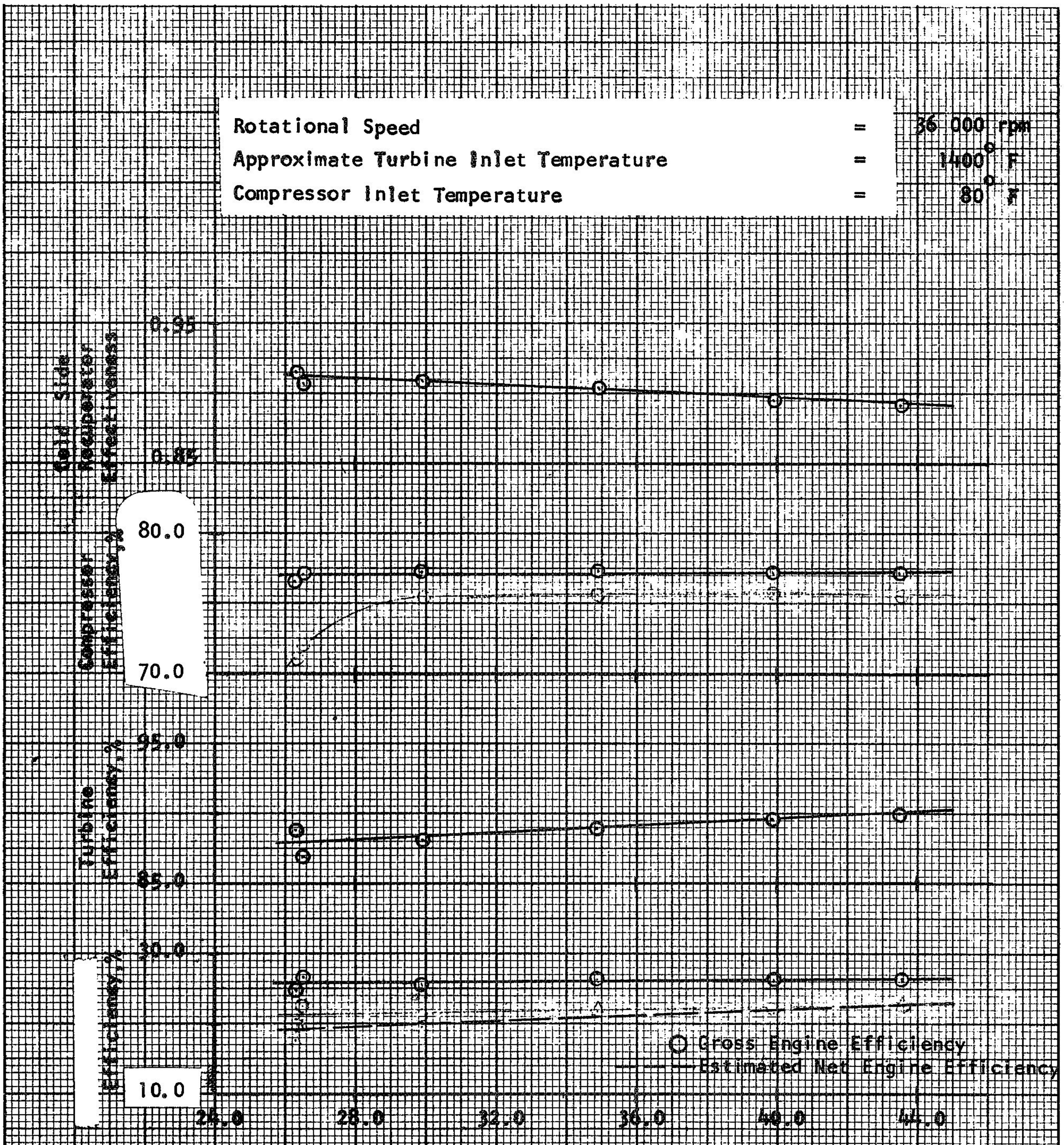

Figure 12. Variation of Engine Efficiencies with Compressor Discharge Pressure

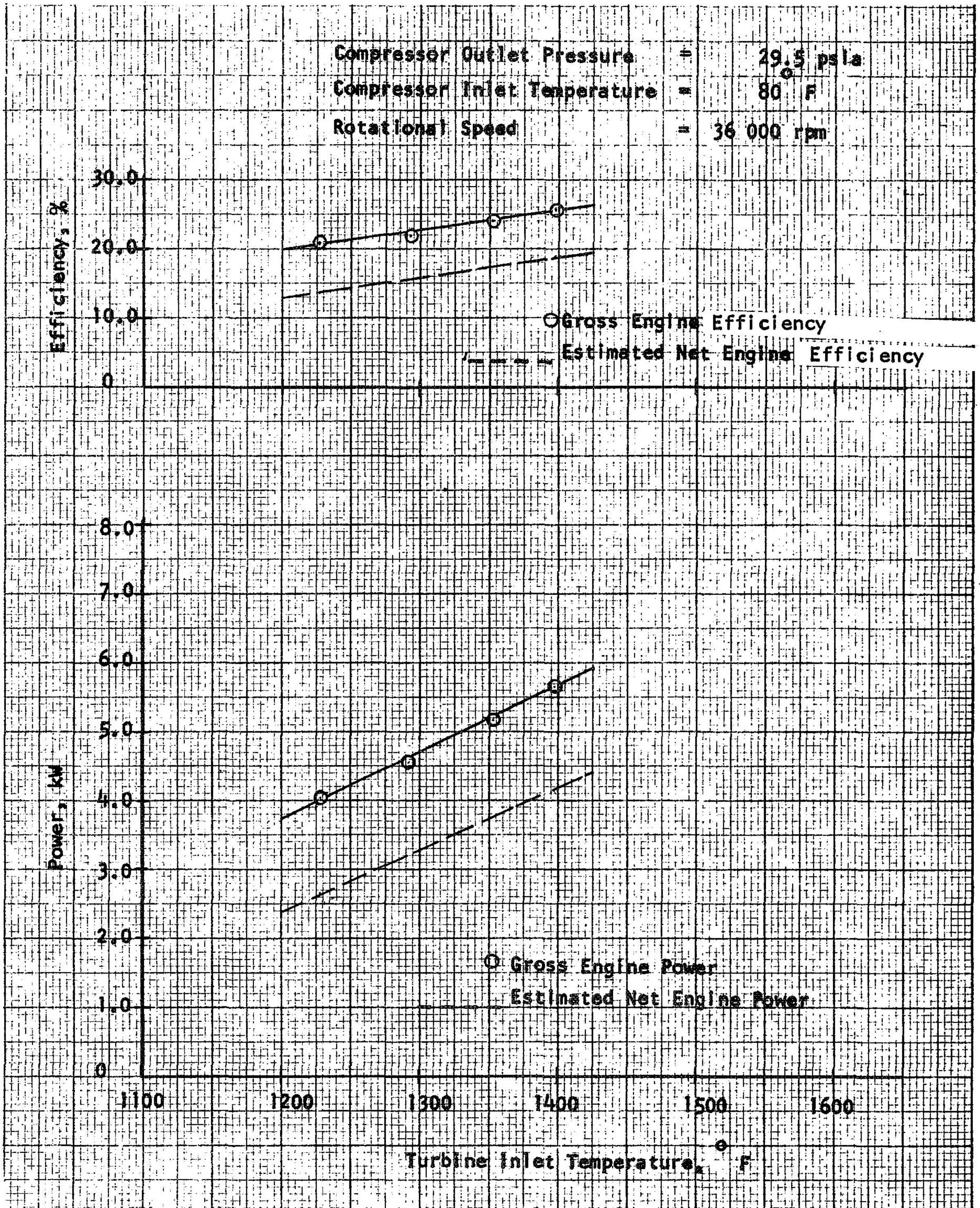

Figure 13. Variation of Engine Performance with Turbine Inlet Temperature

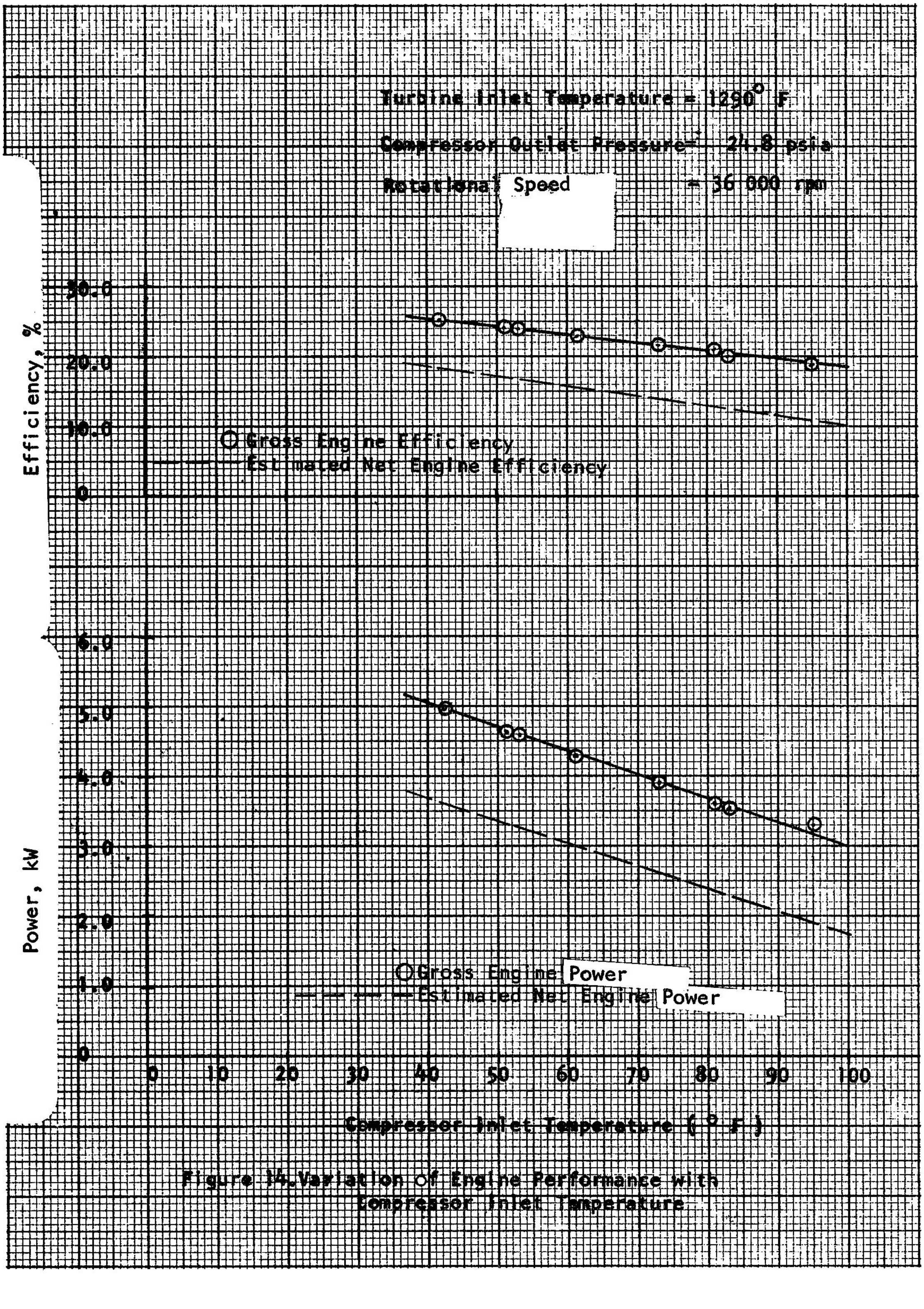

Figure 14. Variation of Engine Performance with Compressor Inlet Temperature